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Mohawk Innovative Technology, Inc.

10,812,281

January 10, 2007

Mail Stop Post Issue
Commissioner for Patents
PO Box 1450
Alexandria VA 22313-1450

Dear Sir:

As per a conversation with your office, please consider this letter a formal complaint with regard to patent #7,108,488 B2, "Turbocharger with Hydrodynamic Foil Bearings", assigned to Honeywell International, Inc.

The technology involved in this patent has been in presented and published literature belonging to Mohawk Innovative Technology, Inc. since at least 2000. For example:

- "Oil-Free Bearings Turn at Highest Speeds", an article featured in European Automotive Design in February 1998.
- "Oil-Free Turbocharger Demonstration Paves Way to Gas Turbine Engine Applications", the featured topic in Mohawk Innovative Technology, Inc.'s newsletter, MiTi® Developments, Volume 6, Spring 1999.
- "Oil-Free Turbocharger Demonstration Paves Way to Gas Turbine Engine Applications", Paper 2000-GT-0620, a presentation given at the ASME Turbo Expo, Land, Sea and Air, IGTI in Munich, Germany in May 2000.

With all the information, expertise and intimate knowledge Mohawk Innovative Technology, Inc. has regarding Oil-Free Turbocharger research activity and after being funded through the Department of Energy, why was our company's prior art not disclosed nor our work or ideas cited? Caterpillar Corporation, who partnered in the development with Mohawk Innovative Technology, Inc. and was the end user of our technology, had been in contact on many occasions with the Turbocharger Division of Honeywell to encourage them to conduct research of their own and had disclosed our joint, ongoing research to them. We have presented numerous papers and articles in the United States, Europe and the Pacific Rim at conferences generally attended by engineers representing Honeywell as well. How is it that Honeywell was awarded this patent? A reply from your office with regard to this matter is anticipated.

Sincerely,

H. Heshmat

Hooshang Heshmat, Ph.D.
President and CEO/Technical Director

HH/cmj
cc: James C. Simmons, Esq.

Encs.

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Proceedings of TURBOEXPO2000:
ASME TURBO EXPO 2000
May 8-11, 2000, Munich, Germany

2000-GT-620

Oil-Free Turbocharger Demonstration Paves Way to Gas Turbine Engine Applications

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Christopher Della Corte, PhD. **Mark Valco, PhD.**

ABSTRACT

An oil-free, 150 Hp turbocharger was successfully operated to 100% speed (95,000 rpm), with turbine inlet temperatures to 650°C on a turbocharger gas test stand. Development of this high speed turbomachine included bearing and lubricant component development tests, rotor-bearing dynamic simulator qualification and gas stand tests of the assembled turbocharger. Self acting, compliant foil hydrodynamic air bearings capable of sustained operation at 650°C and maximum loads to 750 N were used in conjunction with a newly designed shaft and system center housing. Gas stand and simulator test results revealed stable bearing temperatures, low rotor vibrations, good shock tolerance and the ability of the rotor bearing system to sustain overspeed conditions to 121,500 rpm. Bearing component development tests demonstrated 100,000 start stop cycles at 650°C with a newly developed solid film lubricant coating. In a separate demonstration of a 100 mm compliant foil bearing, loads approaching 4,500 N were supported by a compliant foil bearing. This combination of component and integrated rotor-bearing system technology demonstrations addresses many of the issues associated with application of compliant foil bearings to gas turbine engines.

INTRODUCTION

Advanced high performance diesel and gas turbine engines being developed for heavy duty automotive and general aviation transportation applications will operate at higher temperatures, speeds and pressures than ever before as efforts to enhance efficiency and reduce pollutants are pursued. These conditions place considerable stress on the bearings and lubricant supply systems, which in turn reduces bearing life and reliability. Eliminating the oil lubricated bearings and associated supply system will simplify the rotor system, but will increase internal temperatures requiring bearings capable of operating at temperatures approaching 650 C, and at high speeds and loads. Besides surviving the extreme temperatures and speeds, the oil-free bearings will also need to accommodate the operational maneuver and shock conditions experienced in mobile applications. Newly emerging compliant foil bearing technologies and the corresponding high temperature lubricant coatings have made development of oil-free turbochargers and gas turbine engines possible. Besides the recent advances that address the high load and temperature requirements, the application of compliant foil bearings (CFBs) to turbochargers and gas turbine engines draws on a long and successful application history. CFBs have been applied to air cycle machines [1-9], cryocoolers, cryogenic pumps, turboexpanders and other extreme environment systems [10-14]. From Figure 1 [4], it can be seen that CFBs have been previously investigated for application to gas turbine engines [15-22], turbochargers, and turboalternators [23, 24]. However, limitations in high temperature life and load capacity have relegated their use to light-weight, low-temperature rotor systems until now.

In a series of collaborative tests MiTi and NASA have recently demonstrated compliant foil bearing component performance and life in excess of requirements for both heavy duty automotive and general aviation applications. For example, advanced CFBs

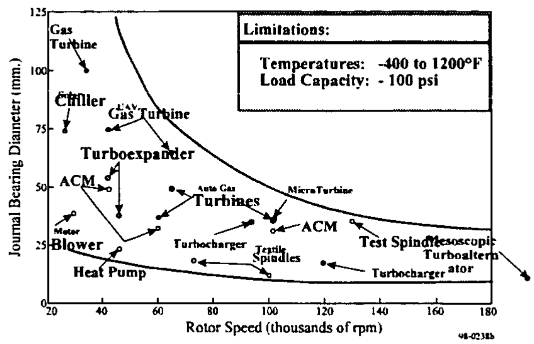

Fig. 1 Spectrum of Foil Bearing Applications

have demonstrated life in excess of 100,000 start stop cycles at 650°C (1200°F). This life exceeds most automotive needs by a factor of three and aerospace applications by a factor of more than five. Speeds in excess of 4.5 Million DN (where D is bearing bore Diameter in mm and N is shaft speed in rpm) have also been demonstrated and a major breakthrough in load performance has also been achieved reaching 674 kPa (97.7 psi) [25], (see Figure 2). The increase in load capacity has also been instrumental in developing the hybrid foil-magnetic bearing, a unique load sharing bearing combination which takes advantage of the strengths of both magnetic and foil bearing elements to make them suitable for application to larger gas turbine engines and rotating machinery [26].

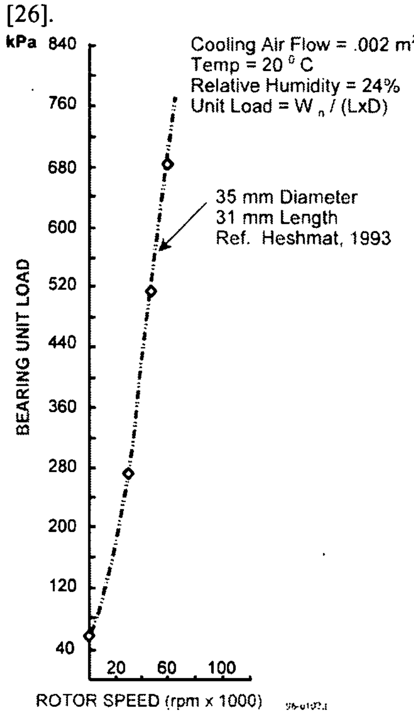

Fig. 2 Load capacity of compliant foil journal bearing

OIL-FREE TURBOCHARGER

The overall purpose of the turbocharger development program was to demonstrate that both heavy duty diesel engine turbochargers and aerospace gas turbine engines can benefit from the use of oil-free compliant foil bearings. The baseline turbocharger used for this demonstration was a Schwitzer 150 HP unit that has a normal maximum operating speed of up to 95,000 rpm and an overspeed limited to 117,000 rpm. The modified turbocharger installed on the gas stand as shown in Figure 3 , incorporated compliant foil journal and thrust bearings (See Figure 4), the PS304 BaF/CaF eutectic lubricant coating at the journal and thrust bearing locations, a new shaft with integral thrust runner and a revised sealing arrangement for thermal management. The rotor and bearing designs were established through iterative rotordynamic, bearing and thermal analyses to ensure that critical speed margin, system stability, bearing load capacity and cooling flows were adequate for safe and durable operation. The resulting shaft and journal bearings are shown in Figure 5 and the design data are presented in Table 1.

Table 1. Rotor and Bearing Design Parameters

Component	Dimensions (mm)	Weight (Newtons)	Design Type	Design Load (Newtons)
Rotor	240 L	20		
Compressor End Journal Bearing	26 L x 36 D		Single Pad	± 750
Turbine End	26 L x 36 D		Single Pad	±
Thrust Bearing	45 ID x 90 OD		8 Pad	± 600

SIMULATOR DEVELOPMENT

Development of this turbocharger followed the procedures established in previous successful system applications, such as air cycle machines and turboexpanders [12, 14]. The fundamental approach begins with a preliminary design tradeoff study and assessment of changes needed to accommodate the compliant foil air bearings. This is followed by a detailed dynamic analysis and design of the selected rotor-bearing system configuration. Based on this system analysis, a dynamic simulator is then designed and fabricated for use in verification testing under precisely controlled conditions. The dynamic simulator is designed to have the same inertia characteristics as the final rotor system, but permits testing under controlled thrust loads and with or without cooling passage seals installed. Additionally, simulators are designed to permit easy hardware changes for rapid testing.

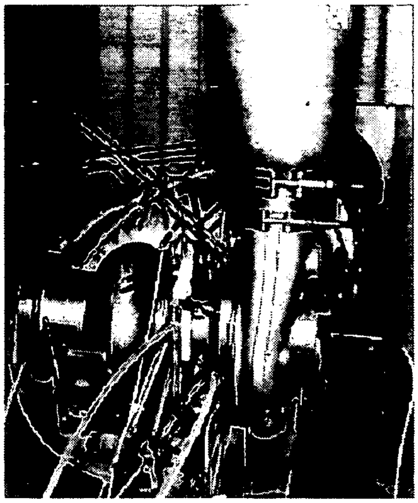

Fig. 3. Oil-Free turbocharger installed on gas stand.

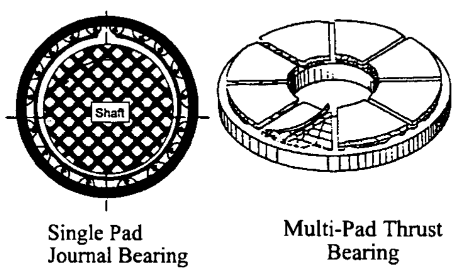

Fig. 4. Compliant foil journal and thrust bearings

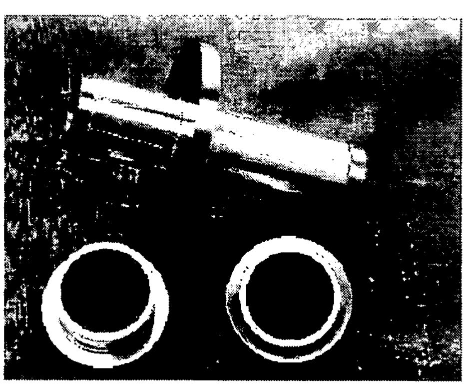

Fig. 5. Turbocharger shaft and journal bearings

The developed finite element rotor model of the turbocharger simulator is shown in Figure 6. The corresponding rotor mode shapes and critical speed map are presented in Figures 7 and 8. As seen the turbocharger is designed to operate above the first and second rigid body critical speeds of 6250 and 8225 rpm and below the first bending critical speed of 167,500 rpm with more than a 20% safety margin.

Besides the rotor system dynamic analysis, a finite element thermal analysis of the turbocharger system was also completed. Figure 9 shows the results of the thermal analysis. The assumed operating conditions were as follows: a 790°C turbine inlet temperature; 270°C cooling flows of .472 L/S through the bearings; and an assumed power loss of 0.225 kW per journal bearing and 0.75 kW per thrust bearing when operating at 110,000 rpm.

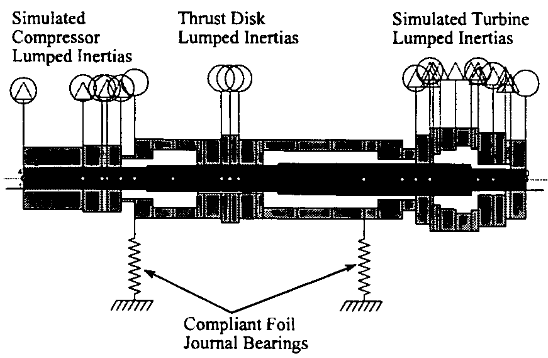

Fig. 6. Turbocharger simulator finite element rotor model.

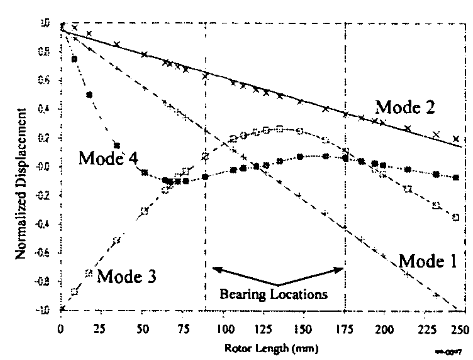

Fig. 7. Rotor undamped mode shapes for first four critical speeds modes.

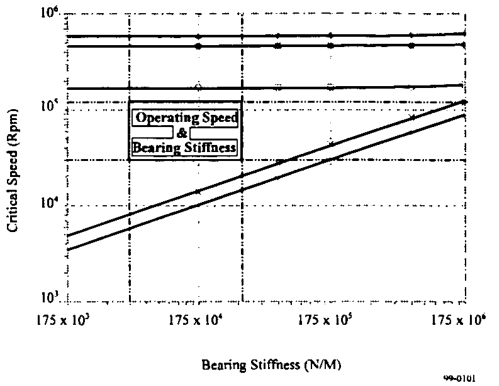

Fig. 8. Critical speed map of turbocharger system.

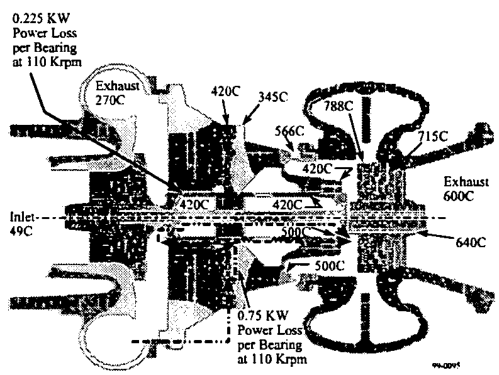

Fig. 9. Thermal map of turbocharger for operation at 110,000 rpm.

SIMULATOR TESTING

The simulator shown in Figure 10 was successfully used to demonstrate the dynamic performance of the rotor and bearing system at speeds to 121,5000 rpm, at various rotor roll orientations, under transient shock conditions and at elevated temperatures. Figure 11 is a waterfall plot of rotor vibrations at the compressor end of the rotor during an over speed test to 121,500 rpm (the coast down is only shown from 120,000 rpm). Peak compressor rotor vibrations during the high speed test were less than approximately 25 μm. This over speed test was used to validate the structural integrity of the shaft which experiences a maximum tip speed in excess of 580 m/sec at the periphery of the thrust runner disk. The over speed test also verified the high speed performance of the bearings.

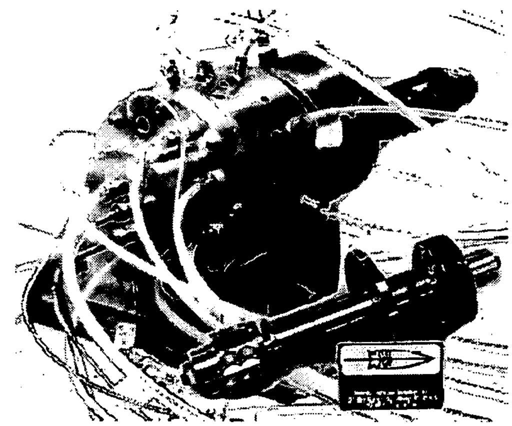

Fig. 10. Turbocharger simulator system showing rotor, simulated drive turbine and compressor wheel.

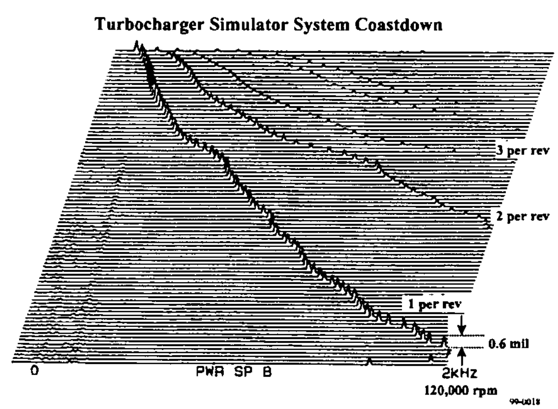

Fig 11. Waterfall plot showing coast down from 120,000 rpm

Following the high speed testing, the simulator was installed on a hinged platform so that a simulated roll type maneuver or condition could be conducted with the rotor operating at a variety of speeds. Roll tests were conducted at speeds to 95,000 rpm and up to 90 degrees roll angle without incident. Once the roll testing was complete, transient shock testing was conducted. In these tests the rotor system was brought up to a selected speed ranging from 60 to 90,000 rpm. The simulator housing was then elevated by a roll maneuver to a specified height and dropped. As seen in Figures 12 and 13, rotor motions for both the 70,000 and 90,000 rpm operating speeds even under shock levels of 19 g, are

not excessive. Rotor motions were limited to a peak to peak amplitude of less than 178 μm. Further these transient vibrations were quickly damped out with rotor orbits remaining at levels less than 25 μm peak to peak. The post test inspection of the rotor and bearings revealed no evidence of shaft to bearing contact.

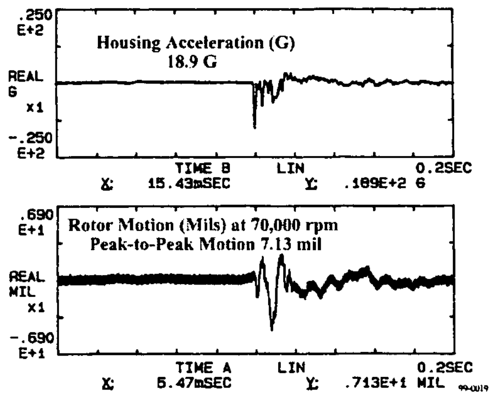

Fig. 12. Rotor response and housing acceleration impact shock for 70,000 rpm operation.

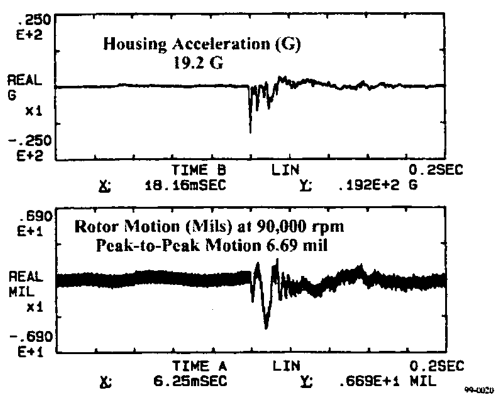

Fig. 13. Rotor response and housing acceleration for impact shock at 90,000 rpm operation.

Following the rotor system dynamic characterization, verification testing of the thrust bearings was accomplished.

TURBOCHARGER TESTING

Following the simulator testing, the rotor with integral thrust runner was mated to the turbocharger turbine and compressor and installed in the turbocharger housing, which was modified to accommodate the new rotor and bearings. This oil-free, 150 Hp turbocharger shown in Figures 3 and 14 was then installed in the Schwitzer turbocharger gas stand and successfully operated to 95,000 rpm (100% speed) with turbine inlet temperatures to 650°C. Figure 15 is a cross section of the oil-free turbocharger showing the location of thermocouples and displacement sensors used during the gas stand testing. Thermal results from testing are shown in Figures 16 and 17. These figures show the temperatures at various locations inside the turbocharger bearing compartments for rotor speeds ranging from 50,000 to 95,000 rpm and with turbine inlet temperatures from 150°C to 650°C. Externally supplied cooling air was used for these first tests in order to evaluate the impact of flow rates and inlet air temperatures. As seen in Figures 16 and 17 the maximum temperature rise over the cooling air occurred in the bearing closest to the turbine and remained below 150°C. Based on this data, finite element thermal analysis and high temperature simulator testing conducted at MiTi, the maximum bearing temperatures are not expected to exceed 540°C in operational units. Vibration measurements also revealed very low rotor displacements as shown in Figure 18. Peak vibrations were approximately 32 μm at the outermost portion of the compressor at the maximum tested speed of 95,000 rpm.

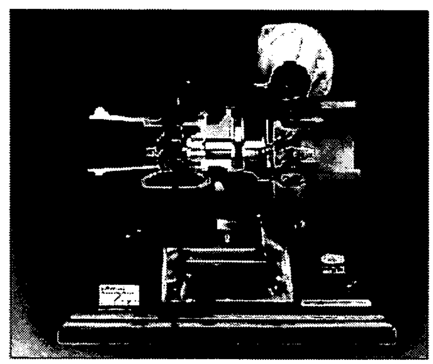

Fig. 14. Cutaway of oil-free turbocharger unit.

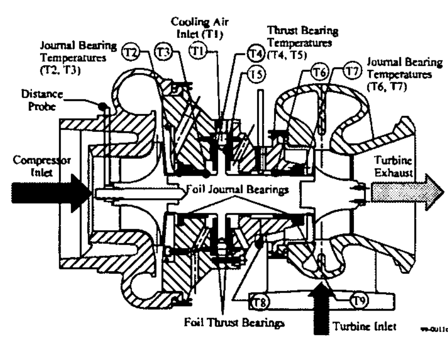

Fig. 15. Turbocharger cross section.

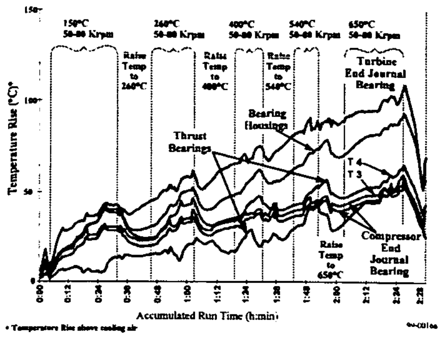

Fig. 16. Bearing temperature rise for turbine inlet temperatures to 650°C and speeds to 80,000 rpm.

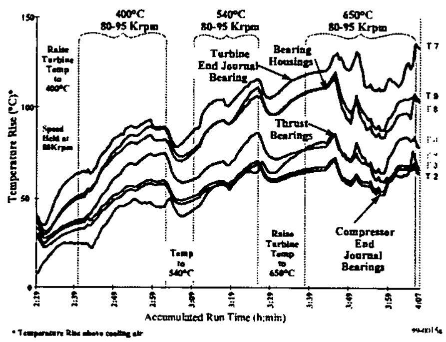

Fig. 17 . Bearing temperature rise above cooling air at turbine inlet temperatures from 400 to 650°C and operating speeds from 80,000 to 95,000 rpm

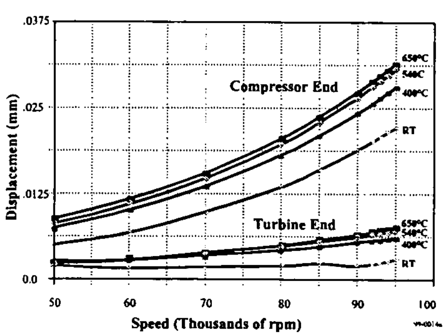

Fig. 18. Turbocharger rotor displacements as a function of speed and temperature

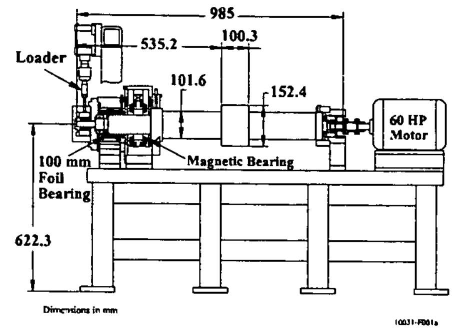

Fig. 19. Simulated turbine engine rotor and foil bearing test rig.

SCALING TESTING

In a parallel development, operation of a 100 mm diameter compliant surface bearing under loads to approximately 4448 Newtons and speeds to 30,000 rpm was demonstrated. This larger bearing is sized to meet a wide range of potential applications such as gas turbine engines for general aviation and helicopter engines as well as larger industrial compressors. The test rig used for this demonstration, shown in Figure 19, consisted of a 60 HP, 36,000 rpm synchronous drive motor, a 985 mm long by approximately 102 mm diameter shaft weighing 625 N. The shaft was supported on the drive end by a damped ball bearing assembly and on the non-drive end by either the 100 mm diameter foil bearing or a 121 mm diameter magnetic bearing. A pneumatically operated loader attached to the end of the shaft through a 35 mm diameter duplex ball bearing pair was used to apply static loads to the shaft at speeds from 10,000 to 30,000 rpm. An actively controlled magnetic bearing was also installed inboard of the foil bearing for use in hybrid foil magnetic bearing testing. The test procedure used for the evaluation of load capacity testing for the 100 mm foil journal bearing was to levitate the rotor with the magnetic bearing, rotate the shaft to the desired test speed, turn the magnetic bearing off, apply static load to the shaft until load reached estimated safe maximum load and then the load was released. As seen in Figure 20 the permissible load capacity measured for the 100 mm diameter foil bearing compare favorably with the data acquired previously for the 35 mm foil journal bearings.

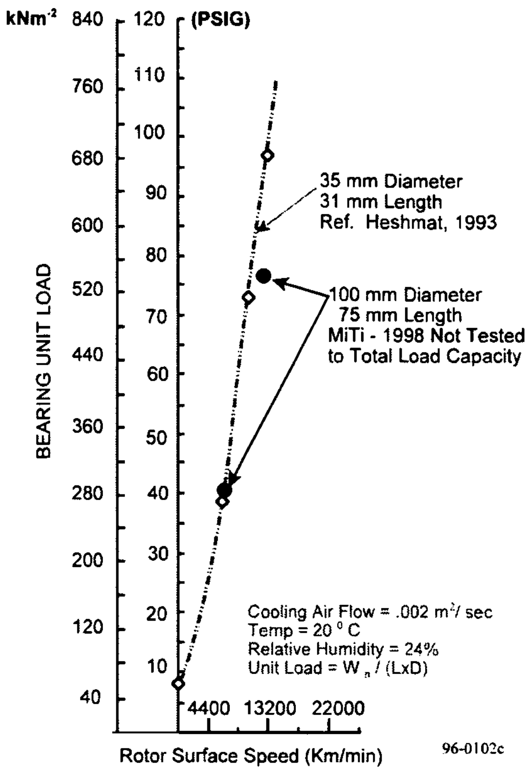

Fig. 20. Comparison of load capacity for 35 and 100 mm diameter foil journal bearings.

SUMMARY AND CONCLUSIONS

This design and test effort has demonstrated the feasibility of developing oil-free turbochargers and small gas turbine engines using compliant foil bearings. This demonstration was accomplished through a methodical development program that has included component, simulator and pre-prototype turbomachinery testing. Component testing demonstrated bearing life to 100,000 start stop cycles at temperatures to 650°C. Component testing with a 100 mm diameter journal bearing at surface speeds comparable to the 35 mm diameter bearings also demonstrated foil bearing load capacity scalability. In both cases the measured load capacity is comparable. These tests demonstrating the ability to scale designs to accommodate absolute loads that are a factor of six to seven times larger than that achieved for state of the art bearings. This increase in total load owes to the corresponding increase in projected bearing area. Through simulator and pre-prototype system testing a turbocharger rotor was operated to speeds in excess of 120,000 rpm, and to 95,000 rpm with turbine inlet temperatures to 650°C. Total bearing temperature rise was less than 150°C even for turbine inlet temperatures of 650°C.

Given the performance of the prototype oil-free turbocharger, demonstrated scalability and total load carrying capability, life and high temperature operation, the barriers to application of foil bearings to a wider range of turbochargers and gas turbine engines have been overcome.

ACKNOWLEDGMENTS

The authors would like to acknowledge the U.S. Army Applied Research Lab, NASA/GRC and Mohawk Innovative Technology, Inc. for their support of this work.

The authors also acknowledge the tireless and dedicated efforts of Mr. Michael Tomaszewski during this effort.

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